HYDRAULIC SHOP PERFORMANCE TESTING OF CENTRIFUGAL COMPRESSORS

by

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ABSTRACT

The purpose of this tutorial is to familiarize the rotating equipment engineer with in-shop hydraulic performance testing methods. The tutorial discusses the theory of similarity testing, the assumptions, and inherent errors. The presentation includes a selected case for demonstration. It is intended for engineers who define performance test requirements for assigned projects, review test agendas, and witness vendor compressor performance tests. Allowable test departures described in the ASME PTC-10 (1997) are reviewed and discussed.

INTRODUCTION

Low pressure inert gas Type 2 (ASME PTC-10, 1997) hydraulic performance testing of a centrifugal compressor is based upon the similitude between the specified condition volume reduction and the test condition volume reduction. Comprehension of volume reduction and the variables that effect volume reduction of the compressor are key to understanding the accuracy and limitations of the test.

For a multistage compressor volume reduction is a critical parameter. The volume reduction of the first stage determines the impeller selection (capacity) of the second stage impeller that, in turn, determines the selection of the following stage and so forth through the discharge of the compressor. Once the final selection is determined the design volume reduction for the rotor is established. In conducting the in-shop performance test, the test volume reduction must match the design volume reduction to be representative of the compressor performance on the specified gas, the objective of the test.

VOLUME REDUCTION

The volume reduction of a compressor stage is dependent upon the polytropic head (work output) and efficiency of the impeller, the gas density, and properties of the medium being compressed. Work applied to the gas through the impeller due to centrifugal force and diffusion results in increased pressure at the discharge of the stage reducing the gas volume. The ratio of discharge volume to inlet volume is the volume reduction of the stage.

To examine the effect of various parameters on the volume reduction of a stage the real gas polytropic head equation is presented.

$$HEAD = \frac{1545}{MW} \times T_1 \times Z_1 \times \left(\frac{n}{n-1}\right) \times \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] \quad (1)$$

$$\left(\frac{n-1}{n}\right) = \left(\frac{k-1}{k} \Big/ efficiency\right) \textit{for an ideal gas} \quad (2)$$

Using Equation (1) assume, for the purpose of demonstration, that all variables remain constant with the exception of head and pressure ratio. If the head is increased, the pressure ratio must increase to maintain equality in the equation. The higher pressure ratio results in increased discharge pressure and lower discharge volume. Increasing head increases the volume reduction; decreasing head decreases volume reduction. Now for example assume that all the variables in Equation (1) are constant except for molecular weight and pressure ratio. If the molecular weight is reduced, the pressure ratio is also reduced to maintain equality. The reduced pressure ratio results in less volume reduction for the stage. Decreasing the molecular weight decreases the volume reduction; increasing the molecular weight increases the volume reduction of the stage.

The same principle may be applied to the other variables in Equations (1) and (2) with reference to the pressure ratio and the following Table 1 developed.

Table 1. Effect on Volume Reduction for Changes in Operating Parameters.

Variable	Change	Effect on Volume Reduction
Head	Increase	Increase
	Decrease	Decrease
Mole Weight	Increase	Increase
	Decrease	Decrease
Inlet Temperature	Increase	Decrease
	Decrease	Increase
Inlet Compressibility	Increase	Decrease
	Decrease	Increase
isentropic exponent (k)	Increase	Decrease
	Decrease	Increase
Inlet Pressure	Increase	No Effect
	Decrease	No Effect

Note that the change in inlet pressure has no effect on the volume reduction of the stage. The discharge pressure changes proportionately to maintain equality; the volume reduction is the same. Pressure effect on the stage volume reduction is limited to the pressure effect on the gas isentropic exponent (k) and the compressibility factor (z).

To visually represent volume reduction the author will use a three-stage compressor for an example. Figure 1 shows three stage curves of flow coefficient (inlet capacity) versus head for the compressor. The design capacity of the first stage impeller is identified by the solid line. The volume reduction of this stage under specified conditions yields the design capacity of the second stage represented by the solid line on the second stage curve. The third stage curve design capacity is established by the volume reduction of the second stage impeller.

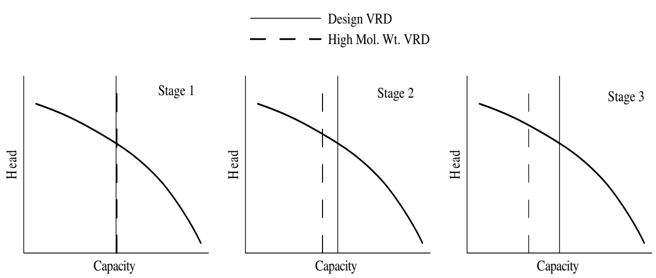

Figure 1. Stage Curve Example for a Three-Stage Compressor.

If the compressor were operated on a molecular weight higher than the specified molecular weight the volume reduction would be greater. For purpose of demonstration assume that only the mole weight is changed. Starting at the design inlet capacity to the first stage, the greater volume reduction due to the increase in molecular weight reduces the operating capacity into the second stage relative to design. At the reduced capacity the second stage produces more head and volume reduction, in addition to the higher mole weight resulting in further deviation from the design capacity at stage three. The operating capacity at each stage at the higher volume reduction due to the higher mole weight is represented as a dashed line on the three stage compressor maps shown in Figure 1.

If this were a test using a higher than design mole weight gas, the volume reduction would be much greater than design. The volume reduction of the compressor would need to be reduced back to design in order to produce a relative performance to the design condition. This is achieved by lowering the head of the compressor relative to design. Therefore, the shop test head will be lower than the design head. The polytropic head of the compressor is compared, test to design, using the polytropic head coefficient.

The polytropic head coefficient (μ) and efficiency of a stage are dependent upon the geometry of the impeller, the capacity being passed, the speed, and the inlet Mach number. The polytropic head of a stage is the product of the head coefficient and the impeller tip speed squared [Equation (3)].

$$Head = \frac{\mu \times U^2}{g} \tag{3}$$

The impeller geometry for the test is the same as design. The head coefficient test to design, by definition, should be equal. Therefore to reduce the head the speed of the compressor is lowered.

For a multistage compressor an overall μ may be calculated using the overall head and the summation of the impeller tip speeds squared [Equation (4)].

$$\mu = \frac{Head \times g}{\sum U^2} \tag{4}$$

The capacity of the compressor is compared during the test using a dimensionless flow coefficient, Q/ND^3. Typically only Q/N is used for the flow coefficient as the impeller diameter is the same test to design.

The head coefficient and efficiency of an impeller vary with the machine Mach number. Machine Mach number is defined as the tip speed of the first stage impeller divided by the inlet sonic velocity of the gas [Equation (5)].

$$Mach\ Number_{machine} = \frac{U}{A_0} = \frac{U}{\sqrt{kgRTZ}} \tag{5}$$

Figure 2 shows a typical stage performance map of head coefficient and efficiency versus inlet flow coefficient at three machine Mach numbers. The variance between the test Mach number and the design Mach number will affect the head coefficient (and volume reduction) of the impeller. Therefore performance testing is required to be conducted at a Mach number relatively close to the design Mach number for each stage and is critical to the accuracy and relevance of the test. For this reason the ASME PTC-10 (1997) provides an allowable departure chart (Figure 3.3 of Code) for differences in the machine Mach number between test and design (Figure 3). The greater the machine Mach number the tighter the allowable tolerance. The Code only addresses the machine Mach number at the first stage, assuming that all following stages have the same percentage departure, test versus specified.

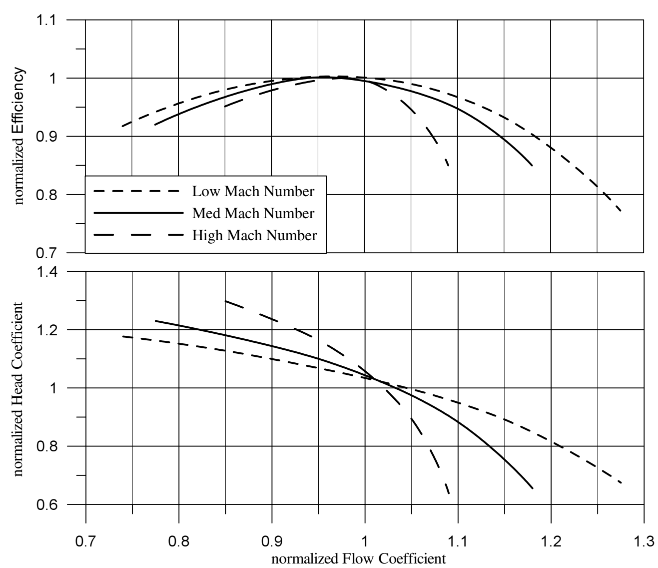

Figure 2. Typical Stage Curve at Three Mach Numbers.

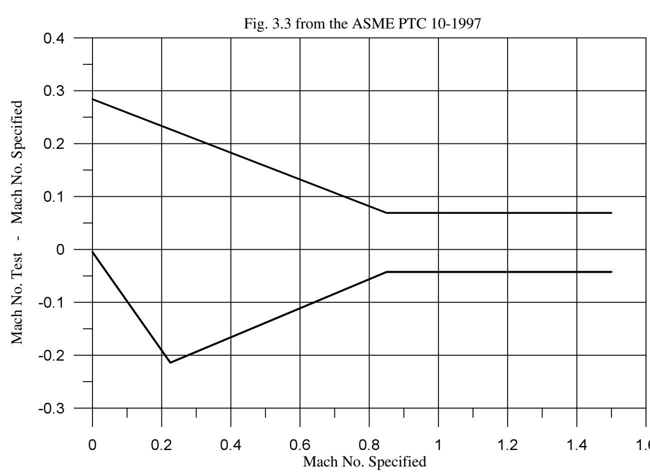

Figure 3. Allowable Test Mach Number Departure from Design. (Courtesy ASME)

DETERMINATION OF TEST CONDITIONS

Of the variables that effect volume reduction the compressor manufacturer has control over the inlet temperature, test gas medium, test speed, and inlet pressure.

The test inlet temperature is dependent upon the cooling capacity of the manufacturers' selected test stand. Typically the test inlet temperature is set at 100°F. The difference in volume reduction caused by a difference in inlet temperature between design and test is corrected via the test speed.

Table 2 lists the typical inert test gases used for an ASME PTC-10 (1997) Type 2 in-shop performance test with the respective mole weights and approximate k values.

Table 2. Typical Inert Gas Mediums for In-Shop Performance Testing.

Gas	Mole Weight	isentropic exponent (k)
Helium	4.01	1.66
Nitrogen	28.01	1.4
Air	28.97	1.4
Carbon Dioxide	44.01	1.26
R134a	102	1.12

Inlet pressure has no effect on volume reduction except for the gas property variance as discussed previously. Inlet pressure does have a significant effect on the test. The pressure directly relates to the amount of mass compressed. The mass determines the power requirement as well as the level of effect the convection and radiation losses to the atmosphere have on the test results. The test Reynolds number is directly proportional to inlet pressure. The effects of heat loss and Reynolds number on the performance test are discussed later. In most cases the test suction pressure may be approximated at 10 percent of the specified value.

A sample compressor application is presented as an example to discuss the selection of the test inlet conditions, gas medium, and speed. The process compressor selected is a five-stage straight-through compressor design. The specified operating conditions are shown in Table 3.

If helium was selected as the test medium for the sample compressor the difference in molecular weight, 4.0 versus 26.3, would result in the test volume reduction to be greatly reduced and therefore the required test speed would exceed the maximum continuous speed of the compressor. For this reason the test gas medium selected is almost always higher in mole weight than the specified gas. A second criterion for selection of the test gas is the k value of the test medium versus the k value of the specified gas. The k value not only affects the volume reduction but also the inlet Mach number. Generally the test gas with the closest k value to the specified gas is selected.

Carbon dioxide was selected for the test gas in the sample compressor. The inlet temperature was set at 100 F. The inlet pressure was set at 44 psia. The test speed to achieve the same overall volume reduction may now be calculated based on thermodynamic law as follows.

Thermodynamic Law:

$$PV^n = constant \tag{6}$$

Therefore:

$$P_1V_1^n = P_2V_2^n \tag{7}$$

Table 3. Specified Operating Conditions.

Inlet pressure	150	psia
Discharge pressure	510	psia
Inlet temperature	110	°F
Discharge temperature	274	°F
Molecular weight	26.3	Mols
Inlet compressibility	0.961	
Inlet capacity	1999	Aft3/min
Mass flowrate	1342.8	lbs/min
Discharge capacity	747.0	Aft3/min
Polytropic head	44,401	ft lbs/lb
Power	2362	gas hp
Bearing loss	52	hp
Speed	13,100	rpm
Machine Mach number	0.658	
Reynolds number	3.63E+06	
Isentropic exponent (k)	1.187	

Rearranging:

$$\left(\frac{P_2}{P_1}\right)^{1/n} = \frac{V_1}{V_2} \tag{8}$$

To have similitude between the specified gas conditions and the test gas conditions, the volume reduction $\left(\frac{V_1}{V_2}\right)$ must be equal.

Therefore:

$$\left(\frac{V_1}{V_2}\right)_{test} \textit{must equal} \left(\frac{V_1}{V_2}\right)_{specified} = \left(\frac{P_2}{P_1}\right)^{1/n}_{test} \tag{9}$$

The polytropic volume exponent (n) is a function of the gas properties and efficiency of the compressor. By definition, the efficiency of the test should equal the efficiency at design. Knowing the test gas properties the test polytropic volume exponent may be calculated and the required test pressure ratio to meet the specified volume reduction may be established.

The test required head is then determined to meet the specified volume reduction using Equation (1).

Head for a compressor is proportional to the speed squared [refer to Equation (3)].

Therefore:

$$Speed_{test} = Speed_{specified} \times \sqrt{\frac{Head_{test}}{Head_{specified}}} \tag{10}$$

With the test gas selected and the test inlet conditions given the test speed was calculated at 11,070 rpm. The overall test conditions at the design point are shown in Table 4.

These conditions are now reviewed and compared to the ASME PTC-10 (1997), Table 3.2, "Allowable Departures from Specified Conditions" for Type 1 and Type 2 tests. The comparison for the sample compressor is given in Table 5.

Table 4. Overall Test Conditions at Design Point.

Inlet pressure	44.0	psia
Discharge pressure	170.2	psia
Inlet temperature	100.0	°F
Discharge temperature	348.9	°F
Molecular weight	44.01	mols
Inlet compressibility	0.987	
Inlet capacity	1799	Aft3/min
Mass flowrate	587.9	lbs/min
Discharge capacity	672.0	Aft3/min
Polytropic head	31,707	ft lbs/lb
Power	743	gas hp
Bearing loss	37	hp
Speed	11,070	rpm
Machine Mach number	0.693	
Reynolds number	1.03E+06	
K value	1.261	

Table 5. Allowable Departures for Sample Compressor at 11,070 RPM.

Parameter	Symbol	Design	Test at 11070 rpm	Test to Design departure ratio or delta	Allowable departures per Table 3.2 ASME PTC 10-1997
Specific Volume Ratio	V1 / V2	2.677	2.766	1.033	0.95 to 1.05
Flow Coefficient (total flow including recycle)	Q/N	0.1625	0.1625	1.000	0.96 to 1.04
Machine Mach Number	Mm	0.658	0.693	0.035	-0.097 to 0.108
Machine Reynolds number	Rem	3.63E+06	1.03E+06	0.284	0.10 to 1.38

There are inherent errors in Type 2 testing based on some of the assumptions taken when calculating the test conditions. The speed calculation is based on the inlet and discharge endpoints assuming a straight line polytropic path on the pressure-enthalpy diagram. The path is truly a slight curve. There are differences in Mach number between the test and specified condition. The allowable Mach number departure between design and test is compared only at the first stage of a section. The departure is not equal at all stages in a multistage compressor. The compressibility factor (z) change from inlet to discharge typically is greater under specified conditions than under the low-pressure test condition. These all cause variations in the individual stage volume reduction from specified. To evaluate the overall effect of the variances, compressor performance at the test condition is calculated on a stage by stage basis. The results are then converted to the dimensionless head coefficient (μ), efficiency, and flow coefficient. This test performance curve is plotted on the predicted curve at specified conditions for comparison. The results from the stage by stage calculation of the sample compressor are presented in Figure 4. The head coefficient versus capacity curve for the test condition is not in agreement with the specified gas curve.

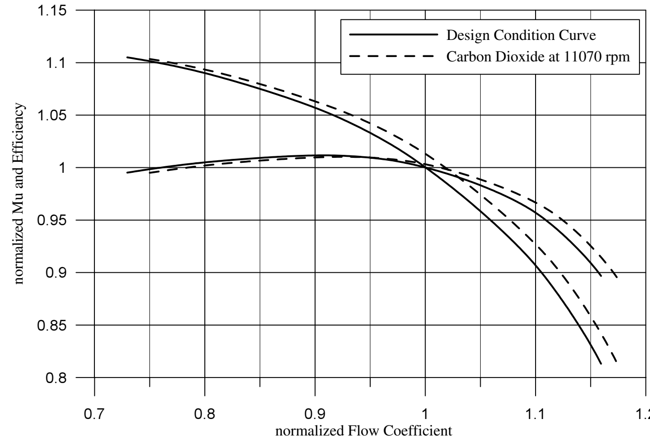

Figure 4. Head Coefficient and Efficiency Versus Inlet Flow Coefficient at 11,070 RPM.

For the majority of applications the predicted stage by stage curve will fall on the specified curve with little variation. The author purposely selected an example where the curves do not match well, even though the test condition is within the allowable departures as given by the ASME PTC-10 (1997) for a Type 2 test.

To visualize what caused the variation at the design inlet flow coefficient the test flow coefficients for each stage were normalized using their respective design values as the reference (Figure 5). If the test is perfect, the test condition line will fall exactly upon the design line in Figure 5. However at 11,070 rpm, the Code established test speed, the first stage volume reduction is greater under the test condition than under specified conditions. This results in the second stage operating at a flow coefficient less than design. As shown earlier if a stage operates at a lower capacity it will generate more head. The additional head and the test variances result in stage two producing a greater volume reduction. This effect continues through the stages until the fourth and fifth stages operate at almost 3 percent less relative capacity than under the specified condition. Overall the volume reduction at the test condition (11,070 rpm) is greater than the design and therefore a poor curve match.

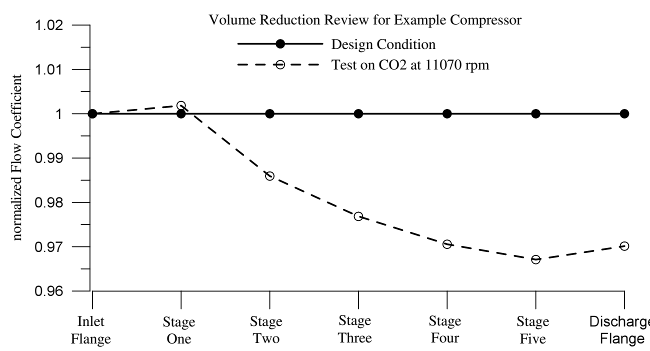

Figure 5. Normalized Flow Coefficient—11,070 RPM.

To correct for the higher volume reduction the test speed of 11,070 rpm was lowered reducing the volume reduction and improving the stage by stage match. A corrected test speed of 10,849 rpm was determined. A second stage by stage calculation was conducted at the revised speed and the predicted curve plotted on the specified gas curve (Figure 6).

The predicted curve for the test is now found to agree closely with the specified gas predicted curve. At the design inlet flow coefficient the stage flow coefficients were normalized and plotted as before (Figure 7). At the lower test speed of 10,849 rpm no stage operates greater than 1 percent from its design flow coefficient.

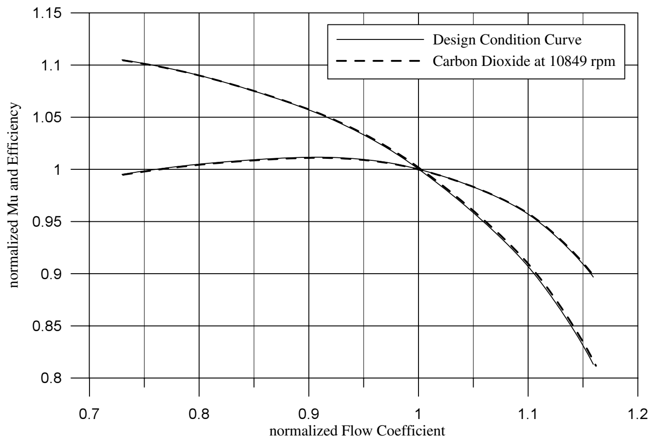

Figure 6. Head Coefficient and Efficiency Versus Inlet Flow Coefficient at 10,849 RPM.

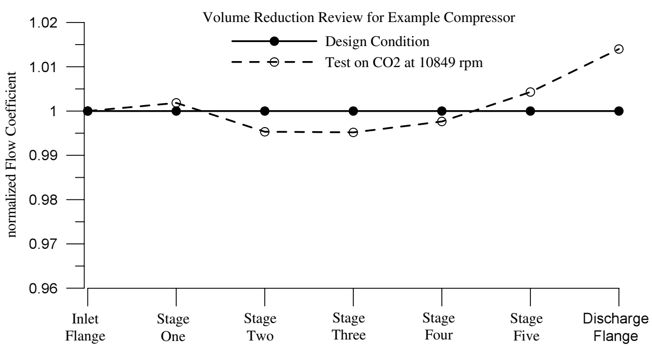

Figure 7. Normalized Flow Coefficient—10,849 RPM.

The stage by stage test condition at the lower speed is again compared to the allowable departure table to ensure that the Code requirements are met. Table 6 yields the departure values from the specified condition for the stage by stage calculations at the Code speed of 11,070 rpm and the adjusted speed of 10,849.

Table 6. Allowable Departures for Sample Compressor at 10,849 RPM.

Parameter	Symbol	Design	Test at 10849 rpm	Test to Design departure ratio or delta	Allowable departures per Table 3.2 ASME PTC 10-1997
Specific Volume Ratio	V1 / V2	2.677	2.645	0.988	0.95 to 1.05
Flow Coefficient (total flow including recycle)	Q/N	0.1625	0.1625	1.000	0.96 to 1.04
Machine Mach Number	Mm	0.658	0.679	0.021	-0.097 to 0.108
Machine Reynolds number	Rem	3.63E+06	1.01E+06	0.278	0.10 to 1.38

The stage by stage calculation of the test condition is a check of the test condition for the combined effect of all variances between test and specified conditions. Requesting a plot of the test curve on the specified curve from the manufacturer will validate that the test condition meets the objective of the test.

DETERMINATION OF SUCTION PRESSURE WITH RESPECT TO AVAILABLE POWER, REYNOLDS NUMBER, AND HEAT LOSS TO ATMOSPHERE

In determining the test suction pressure consideration must be given to the power available in the manufacturer's shop, the Reynolds number variance from specified, and the heat loss through the compressor casing. Inlet pressure is directly related to inlet density and therefore the mass compressed and subsequently the power. The test must be run within the power limits of the available shop driver.

Reynolds number relates to the boundary layer and frictional losses of the medium in the flowpath. Type 2 in-shop performance tests are almost always conducted at a lower Reynolds number than under specified conditions. The lower the Reynolds number the greater the frictional losses and lower the efficiency. This means that the head and efficiency observed during the test will be lower than what would be observed in the field at the same flow coefficient. The ASME PTC-10 (1997) (Figure 3.5) provides a chart showing the allowable departure in test Reynolds number relative to the specified value. The Code also provides a method to calculate a correction to be applied to the test data for the variance in Reynolds number. Utilizing the ASME PTC-10 (1997) method for Reynolds number correction, a graph is presented for the sample compressor (Figure 8). For this sample the allowable range for test Reynolds number is 10 percent to 138 percent of the specified value. If the test was conducted at the minimum 10 percent limit the anticipated correction would be 1.2 percent. Note in Figure 8 that the correction is not linear. The lower the absolute value of Reynolds number the greater the correction exponentially. The minimum allowable Reynolds number absolute value given by the Code is 90,000.

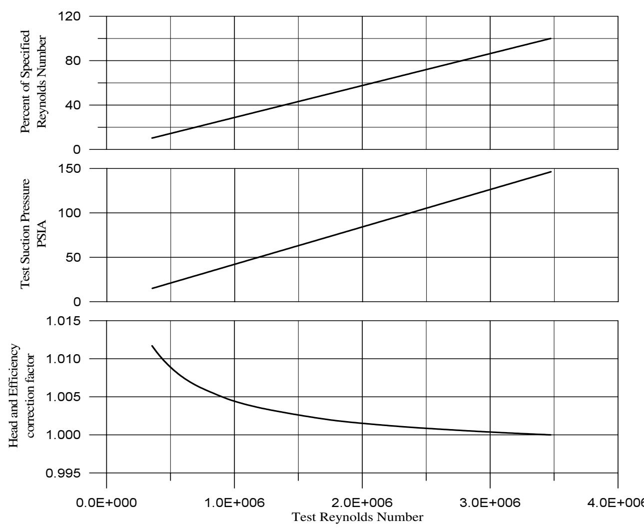

Figure 8. Reynolds Number Correction.

Most users do not allow the manufacturer to correct the test data for the differences in Reynolds number, test to design. The manufacturer should select a suction pressure to maintain the Reynolds number correction at its lowest level within the power and pressure restraints of the test stand. Even if correction is not allowed, it is recommended that the user be aware of the test level.

There are two methods for determining the efficiency of the compressor. A torquemeter may be used to measure the work input into the compressor. From this power the friction of the bearings and seals may be subtracted to obtain the power input into the gas. The head, work output, is divided by the work input to achieve the efficiency. The second (most common) method, is the heat balance method where the work input is measured via the enthalpy rise from inlet to discharge.

Using the heat balance method requires an understanding of the heat lost to atmosphere through the casing by way of conduction and radiation. The heat loss to atmosphere subtracts from the work input measured and if not minimized, eliminated, or accounted for will result in a falsely high efficiency. As the mass throughput increases, the percentage of heat loss to atmosphere relative to the work input decreases. Here again the importance is seen of test suction pressure. The higher the pressure the greater the mass. For the sample compressor the convection and radiation losses were estimated at various test suction pressures and the results were plotted in Figure 9. The 44 psia suction pressure selected for the sample compressor yields an approximate 0.44 percent heat loss to atmosphere, essentially countering the Reynolds number correction. In the few cases where the heat losses cannot be minimized it is recommended that the casing be insulated for the performance test. The user is advised to ask what the approximate heat losses are for the in-shop performance test to ensure that the efficiency observed is representative.

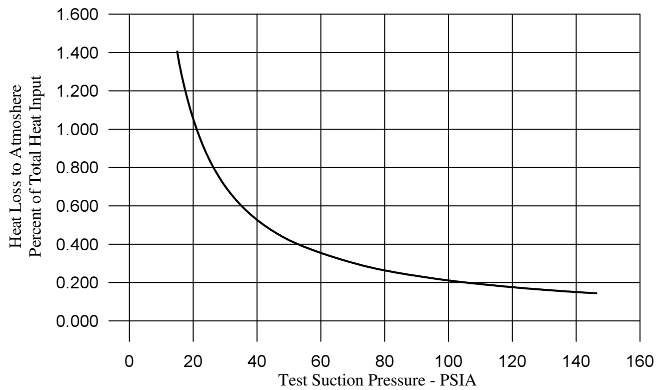

Figure 9. Convection and Radiation Losses to Atmosphere.

MEASURING RECIRCULATION

At the specified operating point API 617, Seventh Edition (2002), states that the horsepower shall be guaranteed within 4 percent of the quoted value. The percentage varies dependent on the user. Note that efficiency is not guaranteed. The power required is the product of the total mass compressed and the work input into the gas [Equation (11)]. It is recommended that where possible, any recirculation or leakage losses should be measured during the shop performance test and the results applied to the specified condition in the conversion of results.

$$GHP = \frac{Head \times Mass}{33000 \times Efficiency} \tag{11}$$

Shown in Figure 10 is a typical loop schematic for a straight-through single section compressor. The straight-through unit loop consists of an inlet flowmeter, a discharge throttle valve, and a heat exchanger. The balance piston return line is piped through a flowmeter and into the test loop upstream of the main inlet flowmeter. This enables measurement of the balance piston seal leakage. The predicted balance leakage is computed similar to an orifice equation. Manufacturers do not all use the same equations for calculating predicted leakages but the physics of the computation is common. A simplified example is given in Equation (12).

$$mass = C \times \Pi \times D \times S \times \sqrt{\frac{(P_h - P_l)}{density_h}} \tag{12}$$

where:

C = Constant
D = Seal diameter
S = Seal clearance
P = Pressure

Subscripts:

h = High side of seal
l = Low side of seal

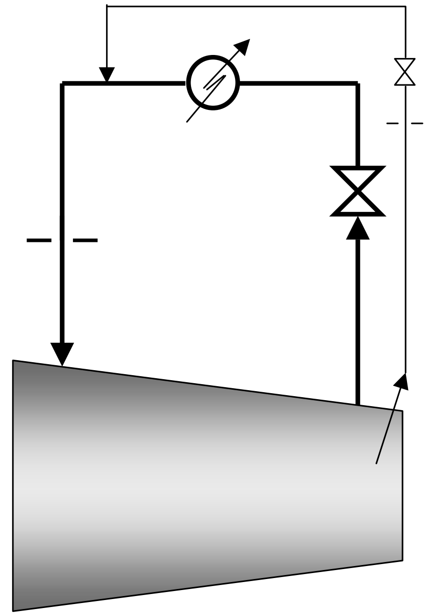

Figure 10. Typical Test Loop—Straight-Through Compressor Design.

During the test the measured value of the balance leakage is known via the flowmeter installed into the return line. The discharge pressure of the compressor may be assumed to be the upstream condition of the seal and the flowmeter upstream pressure assumed as the downstream condition. From these data a pseudo clearance may be calculated and used to convert the results to specified conditions. This is considered a pseudo clearance as the calculated value corrects for any error in the constant.

Back-to-back compressor designs have two major recirculation paths that need to be accounted for in the gas power calculation: the division wall leakage and the end seal leakage. The division wall seal is between the first section discharge and the second section discharge. Leakage from the second section discharge passes across the seal into the first section discharge through the interstage and recompressed by the second section impellers. The end seal leakage (also referred to as the seal balance leakage) passes across the end seal at the section two inlet back to the first section inlet. This leakage is compressed by the first section impellers and then passes through the interstage piping and vessels back into the second section suction. Essentially the division wall seal leakage is recycled around the second section and the end seal leakage is recycled around the first section.

Arrangement of the test loop will allow for the measurement of these leakages so that they may be compared to predicted values and used in the conversion of test results. Figure 11 shows a loop arrangement for a back-to-back compressor that enables the leakages to be measured.

Each section is piped with its own test loop similar to the straight-through example. An orifice is placed in the seal balance line to measure the seal leakage directly. It should be noted here that during the test each end seal is referenced separately for seal supply (oil or gas) purposes. The two loops are connected by a balance line between the first section discharge and the second section inlet, which is also orificed. Figure 10 shows an additional balance line, discharge to discharge, the purpose of which will be discussed later.

During the second section test the seal balance line valve is open as is the discharge to inlet loop balance line valve. The discharge to discharge loop balance line valve is closed. This allows the suction pressure of the second section to be very close to the discharge pressure of the first section enabling leakage across both of the seals in question. To maintain a mass balance in the system,

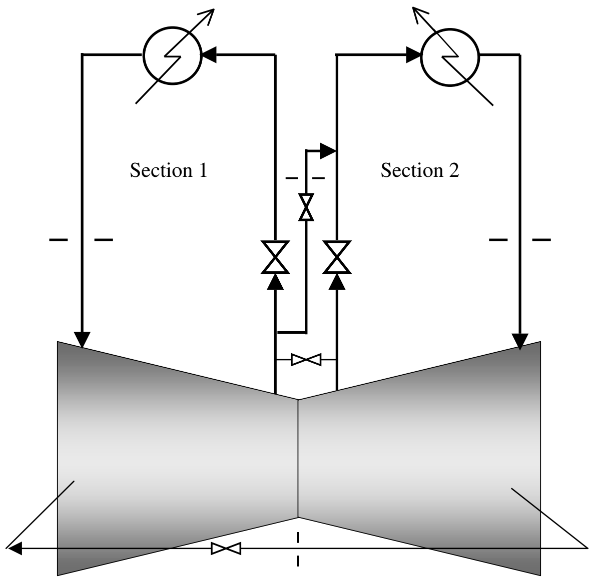

Figure 11. Typical Test Loop—Back-to-Back Compressor Design.

whatever leakage leaves the second section loop to the first section loop must return to the second section loop. It does so via the discharge to inlet loop balance line. The division wall leakage is the difference in mass flow between the loop balance line flowmeter and the end seal flowmeter. The evaluation of the leakage rate is conducted in the same manner as the balance drum leakage. The upstream condition for the division wall is the second section discharge; the downstream condition is the first section discharge. For the end seal, the upstream condition is the second section suction and the downstream condition the first section suction. The flow through the second section impellers is the second section main orifice minus the seal balance orifice.

While testing the first section the valve arrangement must be adjusted to eliminate the recycles so that a proper enthalpy rise may be observed. If the valves were left open then the division wall leakage would mix with the first section discharge mass resulting in an error in the first section discharge temperature observed at the flange. During the section one performance test the discharge to inlet loop balance valve is closed and the discharge to discharge balance line valve opened. The end seal valve is also closed. This arrangement causes the two opposing discharges to be at the same pressure eliminating the division wall seal leakage.

THE STANDARD TEST PROCEDURE

Typically a shop performance test is conducted at only the relative design point speed. The test typically consists of five test points at equally spaced capacities from overload to surge, with one point being at the design inlet flow coefficient. At each flow point the compressor is allowed to reach thermal equilibrium before the data point is recorded. Surge points are also observed at two alternate speeds to define the slope of the surge line. The off-speed surge points are not normally thermally stable before the point is recorded as determining the head and flow coefficient of the point is the objective, not the efficiency. The off-speed surge points also assist the manufacturer in determining where, in which stage, surge was initiated. If the three surge points are very close in flow coefficient, surge is initiated early in the compressor, the first or second stage. A greater separation between the three surge points would indicate surge to be initiated in one of the later stages. Alternate speed points are also observed for constant speed compressors. In this case the slope of the surge line is established for off-design mole weight conditions.

In the past determining surge was done by throttling the compressor until audible surge was detected. Many times this capacity was not the actual surge point of the compressor as observed in the field. Today with the advent of faster data acquisition equipment and vibration monitoring, surge may be accurately determined during the shop test. There are five criteria that may be used to determine the minimum stable flow point.

- Audible surge
- Onset of subsynchronous vibration
- Peak head
- Flow instability
- Pressure instability

Audible flow reversal in the compressor is readily evident as is flow and pressure instabilities. Observing the vibration frequency spectrum is very important in identifying surge. Subsynchronous vibration will occur at the onset of surge providing there is adequate energy input into the unit. The frequency of the vibration generally points to the source of the flow instability. Frequencies in the 6 to 16 percent range may be indicative of a stall in the stationary components of the flow path where frequencies in the 70 to 90 percent range indicate surge is initiated in the impellers. Observing the head coefficient is another method of determining surge. In many cases the head drops off before any other indications are noted. An example is presented in Figure 12. A unit was throttled toward surge slowly with a data acquisition system set to calculate the performance every four seconds. As the unit approached lower flow the head coefficient stopped rising and started to decrease as the flow was reduced further. The head then started to rise again as the flow started to reduce further until an audible surge was found. At the bottom of the head droop, subsynchronous vibration was observed at the journal bearing. The minimum stable flow point was recorded at a capacity corresponding to the initial peak head point. This was at a 10 percent higher capacity than the audible surge capacity.

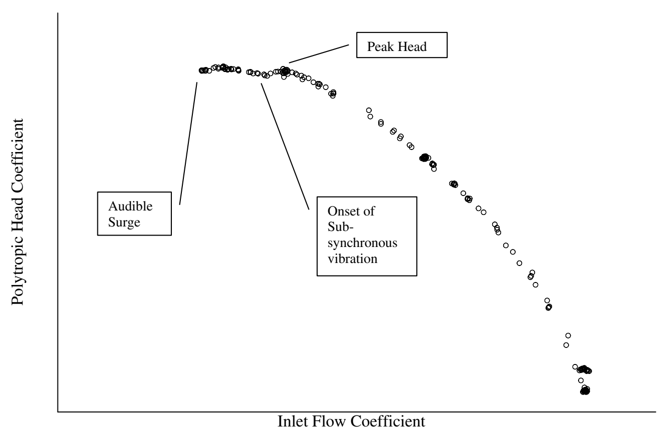

Figure 12. Head Versus Inlet Flow Coefficient—Transition Curve near Surge.

ADDITIONAL SPEED LINE TEST

When should alternate test speed lines, in addition to the specified design point speed line, be requested? It depends upon the compressor, the process, and the user. Discussing the compressor operation with the process engineer will assist the rotating equipment engineer with what are the critical parameters required for the process to run efficiently. If there are alternate operating conditions that are critical to the process the engineer may request a second speed line corresponding to that condition to be tested. It is a matter of risk assessment. The author recommends reviewing the Mach number of the alternate condition with respect to the specified operating point. If the variance is greater than would be allowed by the Code then an additional line may be warranted.

Methods used to extrapolate the overall test performance data established at the specified operating point to alternate operating points varies by manufacturer. A review of the manufacturer's method of extrapolation may also determine whether an additional test at an alternate operating point is warranted.

EVALUATION OF TEST RESULTS

Evaluating the pressure and temperature readings of the performance test to head and efficiency terms should always be conducted using real gas laws and an equation of state for gas properties that is well referenced for the test medium. This tutorial is intended to address the method for establishing proper test conditions for in-shop performance testing. The author recommends the evaluation procedures outlined in ASME PTC-10 (1997) be followed.

TYPE 1 PERFORMANCE TESTING

An ASME PTC-10 (1997) Type 1 test is conducted with the specified gas at or very near the specified operating conditions. Deviations from the specified gas conditions are subject to limitations given in Tables 3.1 and 3.2 of the Code.

The same principles apply to the Type 1 test as to the Type 2 test described earlier. It the test was conducted using the actual specified gas and inlet conditions there would be no deviations from the specified condition to the test. However, this is rarely the case. In many instances the inlet temperature at the specified condition cannot be achieved due to the cooling capacity of the manufacturers' test stand and the specified gas is not available.

The test may be conducted on a gas different from the specified gas; however, the test gas must have a mixture k-value very close to the specified gas. This is required to ensure that the Mach number is the same as specified and that the thermodynamic conversion of the work input to the gas medium produces the same pressure ratio.

Typically the manufacturer has a local gas supply close to pure methane. With this source carbon dioxide, propane, or other gases may be blended to achieve the specified molecular weight and k-value. If the test inlet temperature is greater than the specified inlet temperature the test molecular weight will have to be higher than specified to offset the temperature change and any associated change in compressibility factor. The higher mole weight mixture must still have a mixture k value close to specified.

Reviewing Equation 1 and Equation 5 it can be seen that the product of the gas constant, R (1545/mole weight), inlet temperature, and inlet compressibility are common to volume reduction and machine Mach number. If a blended gas has the same k value and the return-to-zero (RTZ) product is maintained at the inlet, the performance will be the same as on the specified gas. The author recommends that the inlet RTZ product and inlet pressure be maintained within ±2 percent of the specified value. The ASME PTC-10 (1997) (Table 3.2) allows a much greater tolerance.

Comparable plots of compressor performance maps showing polytropic head, efficiency, pressure ratio, and power may be produced for the specified gas and the planned test gas medium. These plots will demonstrate how the compressor performance operating on the test gas blend and conditions conforms to the compressor performance under the specified gas conditions. These curves should be provided in the test agenda.

The Type 1 test point(s) should be established by agreement between the user and the manufacturer dependent upon the objective of the test. It has been the author's experience that this objective is not always well defined in the proposal stages of a project. It is recommended that discussions between the user and the manufacturer concerning the objectives of the test get started as early in the project as possible. This will ensure that the test stand loop design will accommodate all of the objectives.

SUMMARY

The information provided in this tutorial should provide the process and rotating equipment engineer with a better understanding of performance testing methods for centrifugal compressors. When reviewing the ASME PTC-10 (1997) the engineer will have a greater understanding of why the limitations outlined in the Code are given.

NOMENCLATURE

MW	=	Molecular weight, mols
T	=	Temperature, degrees Fahrenheit
Z	=	Compressibility factor, dimensionless
P	=	Pressure, pounds force per square inch
n	=	Polytropic volume exponent, dimensionless
k	=	Isentropic exponent
U	=	Tip speed, feet per second
μ	=	Polytropic head coefficient, dimensionless
A0	=	Sonic velocity, feet per second
g	=	Gravitational constant, feet per second squared
V	=	Specific volume, cubic feet per pound mass
N	=	Speed, revolutions per minute
Q	=	Capacity, cubic feet per minute
GHP	=	Gas horsepower
Mass	=	Mass flow rate, pounds per minute
D	=	Diameter, inches
S	=	Radial clearance, inches
Density	=	Pounds mass per cubic foot
C	=	A given constant

Subscripts:

1	=	Inlet
2	=	Discharge
h	=	High pressure side (upstream)
l	=	Low pressure side (downstream)

REFERENCES

API Standard 617, 2002, "Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services," Seventh Edition, American Petroleum Institute, Washington, D.C.

ASME PTC-10, 1997, "Performance Test Code on Compressors and Exhausters," American Society of Mechanical Engineers, New York, New York.